HEAT TRANSFER IN TURBULENT FLOW OF FLUIDS IN SQUARE DUCTS

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The fluid dynamics in rectangular ducts have been extensively analyzed in several publications [1-3]. The problems of heat transfer in turbulent fluid flow have not been studied to such an extent. While publications on the flow of fluids in rectangular ducts describe studies not only of the average, but also of local characteristics of turbulent flow (distribution of velocities, turbulent stresses), then in the majority of publications devoted to the study of heat transfer in rectangular ducts present data only on average heat transfer coefficients [4-8]. /82

The boundary conditions in many of these studies were close to conditions of constant wall temperature along the duct perimeter. To calculate the heat transferred to the heat transfer agents with $Pr \geq 1$ it is recommended to use the same formulas which are used for heat transfer in round pipes, and to use the hydraulic diameter as the characteristic dimension. However, this method is not always justified. As was shown in [9-11], the use of the hydraulic diameter does not yield satisfactory results for liquid-metal heat-transfer agents. For example, for liquid metals flowing in rectangular ducts the heat transfer coefficients calculated on the basis of the hydraulic diameter become smaller by almost two-fold, if the duct is heated from one side rather than from two. In calculating high thermal flux heat exchangers it is necessary to know not only the average heat transfer coefficients, but also the local distribution of heat fluxes and temperatures on the heat conducting surface and in the duct walls [11, 12].

Below are presented results obtained in the study of heat transfer in a square duct in turbulent flow of water and mercury. We have studied the temperature fields in the wall and in the fluid, as well as the statistical characteristics of turbulent temperature fluctuations.

Methods Used in the Experiments

Experiments with heat transfer in water were performed in an open-circuit setup. The water flow rate was kept constant by using a 125-liter supply tank. It was provided by a filter for removing impurities. The rate of flow of water in the circuit was adjusted by a valve and measured by an RS-7 rotameter. The experiments with mercury were performed in a closed circuit, with the mercury circulated through it by a centrifugal pump. The mercury flow rate was adjusted by a valve and was measured by a Venturi tube used in combination with the DT-50 differential manometer. Brand R-3 mercury, which was filtered before being circulated, was used in the experiment. /83

The test section consisted of a square duct, installed in the circuit in the vertical position, with a cross section of 30 x 30 mm, wall thickness of 6 mm and 1800 mm long (Fig. 1). The experimental section had at its inlet a chamber for equalizing the velocity distribution, while at the outlet it was provided by a mixing chamber. In both chambers were installed thermocouples in sleeves, used for measuring the temperature of the fluid at the inlet and outlet from the test section. The outlet chamber was provided by a probe for measuring the temperature in the fluid flow. The inside surface of the duct was machined to finish class $\nabla 5$. The distance between walls 2 and 4 (see Fig. 1b) was $30^{+0.19}_{-0.08}$ mm, while the distance between walls 1 and 3 was $30^{+0.34}_{-0.10}$ mm. Wall 4 was used as the basic wall for measuring the temperature distribution.

The heat was supplied to the test section by four electric heaters of the plate type, made from 30 mm wide 1Kh18N9T steel, conforming to the width of the flow cross section of the duct, 0.1 mm thick and 1100 mm long (see Fig. 1b). Each heater was fastened to a square textolite rod and was pressed to the duct by clamp screws, fastened in steel columns. A layer of mica about 0.1 mm thick was placed between the heater and the duct wall. The current and voltage at the heaters were measured by class 0.5 astatic instruments. The heaters could be started up in any combinations. The experiments were performed with the duct heated from four sides, with a heater corner (walls 1 and 4), with heating two opposite walls (walls 2 and 4) and with heating one wall (wall 4).

The temperature distribution in wall 4 was measured at three different distances from the point where heating started: 1000, 1020 and 1040 mm (which corresponded to distances from the inlet of 1645, 1665 and 1685 mm) by 27 microthermocouples, 9 of which were built into each of the cross sections (see Fig.1c). These thermocouples were made from copper and constantan wires 0.09 and 0.10 mm in diameter, coated by Viniflex insulation and placed over the external side of wall 4 into slots 0.1 mm wide and 0.1-0.15 mm deep at an angle of 75° to the duct axis. The hot junction of the thermocouples were welded to the bottom of the slot in such a manner that the junctions of all the nine thermocouples would be situated in a plane perpendicular to the duct axis. The wall temperature was measured differentially, referring it to the temperature of the heat transfer agent leaving the duct. The thermocouple readings were measured by ar R-2/1 potentiometer with a sensitivity limit of 0.01 millivolts. The temperature distribution in the wall of the square duct was measured under steady conditions in the zone of stabilized heat transfer. The distance from the duct inlet to the second cross section (see Fig. 1) is $l/d_h = 56$, while from the start of the heated section it is /85 $l/d_h = 34$.

The temperature in the water and mercury flows were measured by a thermocouple probe (Fig. 2). The copper-constantan thermocouple was made from wires 0.09 and 0.10 mm in diameter with Viniflex insulation, welded together with the capillary. The 0.25 mm diameter thermocouple which is obtained in this manner was placed at a distance of 5 mm from the edge of the cone. The probe was moved along the duct cross section by means of two mutually perpendicular micrometer screws, which made it possible to position the probe at any point in the duct with a precision to within 0.02 mm. In order to reduce the effect

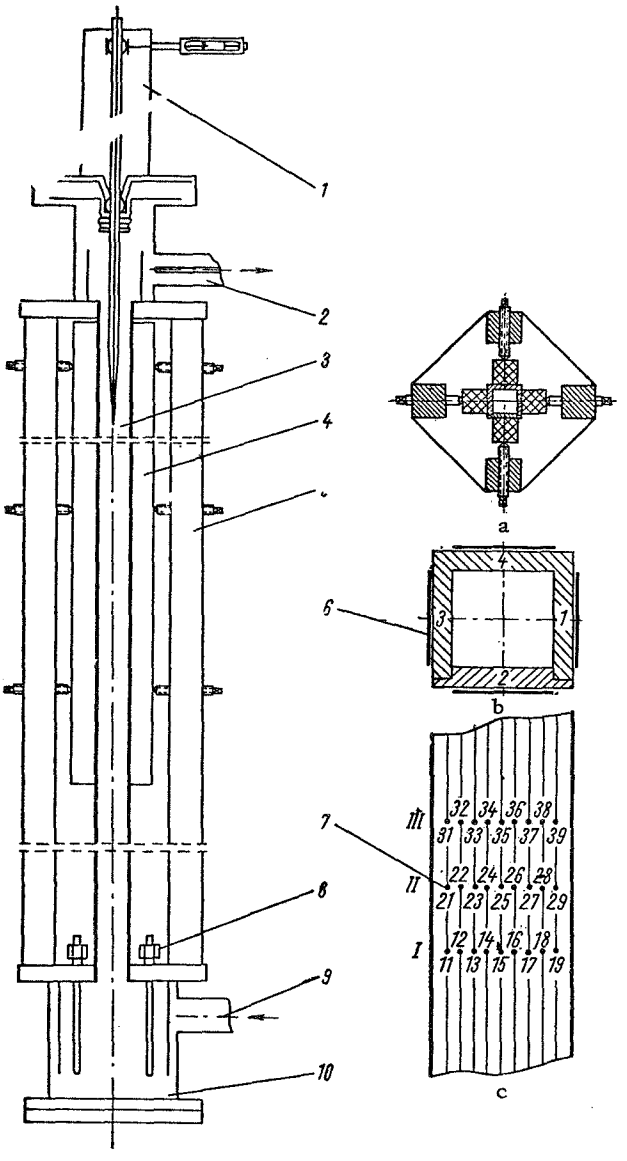


Figure 1. Schematic Diagram of the Test Section. 1) Thermocouple Probe; 2) Outlet Pipe; 3) Square Duct; 4) Textolite Bar; 5) Column; 6) Electric Heater; 7) Thermocouples Built Into the Wall; 8) Inlet Thermocouple; 9) Inlet Pipe; 10) Inlet Chamber. a) Cross Section of Test Section; b) Cross Section of Square Duct (1, 2, 3 and 4 are the Numbers of the Walls); c) Location of Hot Junctions of Thermocouples at the Outer Surface of the Duct (I, II and III are Sections in Which the Termocouples Have Been Inserted); 11-19; 21-29 and 31-39 are Thermocouples Numbers.

of free play, the probe in our experiments was moved in one direction only, i.e., from the center of the duct toward the walls.

The temperature in the water and mercury flows was measured under steady conditions in the region with thermal and fluid flow stability. The moving thermocouple was connected differentially with a thermocouple placed at the outlet from the test section. To obtain average temperatures in each point of the flow, the thermocouple readings were continuously recorded on the diagram tape of the EPP-09 potentiometer with a scale of 0-1 mv during 40-60 secs. The PPTN-1 low-resistance potentiometer, which provided the compensating voltage was installed in series with the EPP-09. The readings of the thermocouple which were obtained for a given probe position were average with respect to time, and these quantities were used to plot graphs of temperature distributions in the flow in various planes along lines perpendicular to the wall. The primary temperature used for analysis of experimental data was the mixed average temperature of the flow in the cross section in which the movable thermocouple moved. The physical parameters of the water were those in [13], while for mercury the parameters of [14] were used.

When measuring the temperature fluctuations the thermocouple signal was fed to an amplifier with a transmission band from 0.18 to

200 cps and an amplification factor of 10^5, equipped with a noise silencer (damping factor of 10^4). The amplified signal was fed either to the MPO-2 loop oscillograph, or to the 46K1 correlometer [15]. The latter could be used to measure the root-mean-square value of temperature fluctuations, the averaging time of which is 5 mins. In addition, the thermocouple readings were recorded by the loop oscillograph on a motion picture film. The time marks were recorded simultaneously on the film. The thermocouple readings were recorded for 60-70 secs in each experiment.

The Heat Transfer and the Temperature Distribution in the Wall of a Square Duct

The results obtained by measuring the temperature distribution at the outer surface of the wall of a square duct are presented in Figs. 3a and b. As can be seen, the temperature of the outer surface of a heat conducting wall is not constant along the perimeter. The temperature distribution when the duct is heated from four sides has the form of a symmetrical curve with two maxima, situated at a distance of about 0.7 y/a from the center line of the duct wall. The shape of the distributions we have obtained depends, naturally, on the velocity distribution in the duct, on the shape, dimensions and thermal conductivity of the duct wall.

It can also be seen from Fig. 3 that, when the duct is heated from four sides, there are observed on the outer surface of the wall temperature minima in zones /88
directly adjoining the duct's corners. These minima are produced by the fact that the heaters are narrower than the outer duct surface [being heated]. The fact that a non-heated, heat conducting section was situated at the end boundary of the heater reduced the thermal resistance of the corner zone and, consequently, resulted in dropping the temperature of the outer surface of the duct. The temperature distribution on the inner, heat conducting wall of the duct, when heated from four sides, will not be identical with the distribution shown in Figs. 3a and b. The temperature maxima at the outer surface for the same amount of heating will be in the corners of the duct. This was substantiated experimentally by measuring the temperature fields in the fluid in the same square duct.

We note that the temperature distributions obtained in the three cross sections (see Fig. 3a) are practically identical with one another. The small differences between them and the slight asymmetricity can be ascribed to bending or displacement of the heater's band, as well as to the fact that it was not uniformly pressed to the duct. The shape of temperature curves on heating from one and two sides is highly affected by the heat flowing along the side, nonheated walls, which in these cases act as cooling ribs.

Curves obtained on heating from two and one side, shown in Figs. 3a and b have a temperature minimum in the corner of the duct. In the given duct the lateral, unheated wall had a relatively high thermal conductivity (9 ~ 10 watts/m-degree). But, as is shown in [12], even if the lateral wall has a thermal conductivity

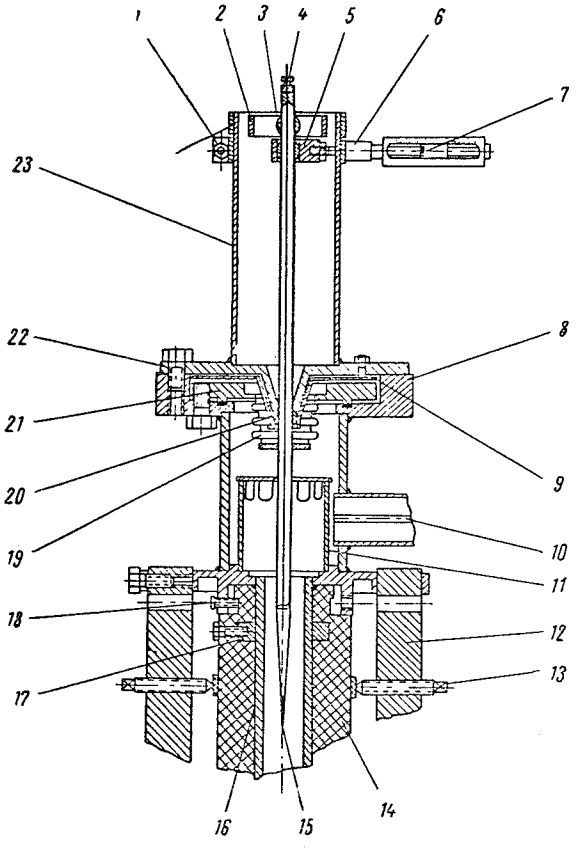


Figure 2. Construction of the Thermocouple Probe. 1) Clamping Ring; 2) X-axis Guide; 3) Ball; 4) Packing Gland; 5) Y-axis Guide; 6) Sleeve; 7) Micrometer Screw; 8) Mixing Chamber Housing; 9) Clamping Cone; 10) Thermocouple Well; 11) Mixing Unit; 12) Column; 13) Heater-Clamping Screw; 14) Textolite Bar; 15) Tube With Thermocouple; 16) Heater; 17) Current-Carrying Heater End Pieces; 18) Adjusting Screws; 19) Bellows-Type Packing; 20) Hinge; 21) Air-Tight Flange; 22) Top Flange; 23) Cup.

of ~ 0.1 watts/m-degrees, the thermal flux to it is so appreciable, that the temperature of the heated wall in a corner of a duct heated from two or one side is lower than its temperature in the central region. The shape of Curve 2 (see Fig. 3a), drawn for heating the corner, reflects the peculiar features of Curves 1 and 3 and is intermediate between them. Curve 2 shows to what distance along the duct's perimeter the effect of these heat leakages on the wall temperature extends.

Up to now we have considered only the shape of the temperature curves. We now compare these curves with the average mean temperature differences, which can be done to estimate the effect of the liquid's thermal conductivity on the heat transfer. For the simplest case, when the fluid flows between two parallel plates, it was shown by Seban [10] that the heat transfer coefficient will be affected by the heating method (i.e., whether the duct is heated from one or two sides) for liquids with $Pr \ll 1$; if, however, $Pr \gtrsim 1$, then the heat transfer coefficient is practically independent of the heating method. Experiments on heat transfer in slot-type and rectangular ducts confirm the fact that the number of sides heated has an effect on the heat transfer coefficient in the case of a liquid metal heat transfer agent [11]. This effect of the Prandtl number on the heat transfer coefficient is a result of the fact that in liquid metals the temperature distribution /89 varies smoothly over the duct cross section, and hence, it depends on the heating conditions of the second wall. In ordinary, low-conductivity fluids the entire temperature drop which determines the heat transfer coefficient is concentrated in the layer near the wall and it is almost unaffected by the heating conditions at the other wall. Let us now compare the results obtained with water and mercury heated from one and two sides (see Figs. 3a and b). While Curves 3 and 4 (water) are practically identical, then Curve 4

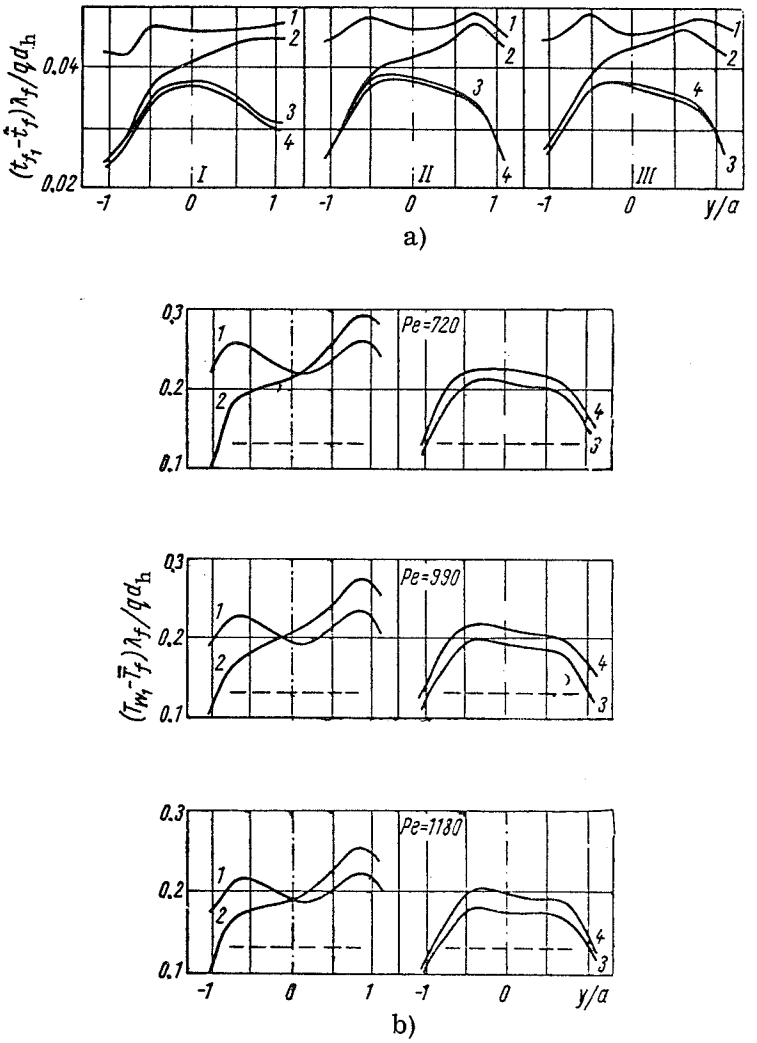


Figure 3. Temperature Distribution on the Outer Surface of a Wall of a Square Duct When Heat is Transferred to Water (Re = 4910) (a) and to Mercury (b). 1) Heating From Four Sides; 2) Heating a Corner; 3) Heating From Two Sides; 4) Heating From One Side. The Dimensionless Distance From the Center Line of the Duct Wall has Been Laid off on the Abscissa Axis, While the Dimensionless Temperature of the Wall has Been Laid off on the Ordinate Axis. The Roman Numerals Denote Numbers of Duct Sections (a). All the Curves of (b) Pertain to the Second Cross Section.

(one sided heating) in experiments with mercury lies noticeably higher than Curve 3. It can be seen from Fig. 3b that the average mean temperature difference for

heating from one side is higher than that for heating from two sides. Fig. 3a shows that, in experiments with water, the heating method does not have this effect on the heat transfer, although in some cases one may observe also a drop in the temperature of the outer, heated wall, produced by increased heat leakage due to the increased effective length of the cooling rib.

Comparison of Curves 1 and 2 (see Figs. 3a and b) shows that, for heat transfer to mercury, the total mean temperature difference in the corner of a duct heated at the corner is higher than for heating from four sides. This increase in the mean temperature difference for two-sided heating as compared with heating from four sides can be ascribed to the same effect of the heating method on the heat transfer as is observed in slot-type ducts. In this case heating from four sides can be regarded as twice heating from two sides. Thus, a reduction in the portion of the heated perimeter on heat transfer in liquid metals increases the mean temperature difference, while the presence of heat flux leakages over the unheated, conducting walls can result in reducing it. In some cases these effects may compensate for one another and the average heat transfer coefficient may not be dependent on that portion of the heated perimeter.

On Fig. 3b are drawn dashed lines the ordinate of which corresponds to the average mean temperature difference over the thickness of the duct wall. The difference between the ordinates of the dashed line and one of the curves for any value of y/a yields an arbitrary value 1/Nu in the given point on the perimeter.

Figures 4a and b show curves of Nusselt numbers as a function of the fluid flow rate attendant to heat transfer to water and mercury. The values of Nusselt numbers were determined from the expression

$$\mathrm{Nu} = \frac{2qa}{\left(\bar{t}_w - \frac{qb}{\lambda_w} - \bar{t}_f\right)\lambda_f}.$$

The Nusselt numbers obtained should be regarded as [somewhat] arbitrary. Figure 4a (line 1-2-3) shows the dependence of Nu on the Reynolds number for nonmetal liquids, which is used for calculating the heat transfer coefficients in round pipes [5]. Segment 1-2 corresponds to transient flow, while Segment 2-3 corresponds to the formula

$$\mathrm{Nu} = 0.021\mathrm{Re}^{0.8}\mathrm{Pr}^{0.43}\left(\mathrm{Pr}_f/\mathrm{Pr}_w\right)^{0.25}.$$

The straight line in Fig. 4b was obtained from a formula used for calculating the heat transfer to liquid metals in round pipes [16] /90

$$\mathrm{Nu} = 5 + 0.025\mathrm{Pe}^{0.8}.$$

The spread of experimental points in Figs. 4a and b can be attributed to two factors: effect of the thermal conductivity of the fluid and of heat flux leakages along the duct's wall on the mean temperature difference for different methods of heating.

Temperature Fields in the Flow of a Fluid

Results obtained by measuring the temperature in the flow of water and mercury are shown in Figs. 5-10. It can be seen from these figures that the temperature distributions for the case of mercury are smoother than for water. As is the case with pipes, the main part of the temperature drop is concentrated in the region near the wall within the limits of $y/a = 0.9-1.0$. As the Reynolds number becomes larger the width of this region is reduced (see Fig. 5a). For mercury the effect of the flow velocity on the temperature distributions at $Re = (16-45)\cdot 10^3$ is weak (see Fig. 5b). /92

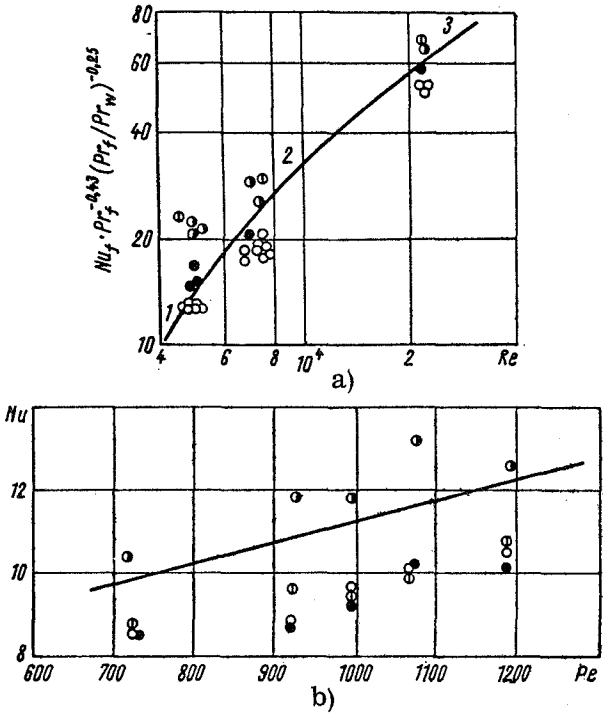


Figure 4. Heat Transfer in a Square Duct to Water (a) and Mercury (b) for Different Heating Conditions. ○ - Heating From Four Sides; ◑ - Heating From Two Sides; ◐ - - Heating From One Side; ● - Heating the Corner.

The shapes of temperature distributions in different duct cross sections for heat transfer to water are not the same. While for the temperature distribution with $x/a = 0$ the minimum is in the center of the duct, then for the remaining distributions the temperature minimum is displaced toward the wall and is situated within the limits of $y/a = 0.6-0.8$ (see Fig. 6a). In heat transfer to mercury the temperature distributions in the [various] duct cross sections have no inflections and the minima are situated at $y/a = 0$. It is remarkable that in the flow region near the wall the slopes of the temperature curves differ appreciably in different cross sections. The maximum slopes of temperature curves in this region in the case of water are observed in the cross section $x/a = 0.90$, and they become smaller as one moved from this cross section to the center of the duct and to the corner region (see Fig. 6a). In the case of mercury the maximum temperature curve slopes are observed in the center zone and become smoothly smaller as one comes closer to the duct's corner (see Fig. 6b). The

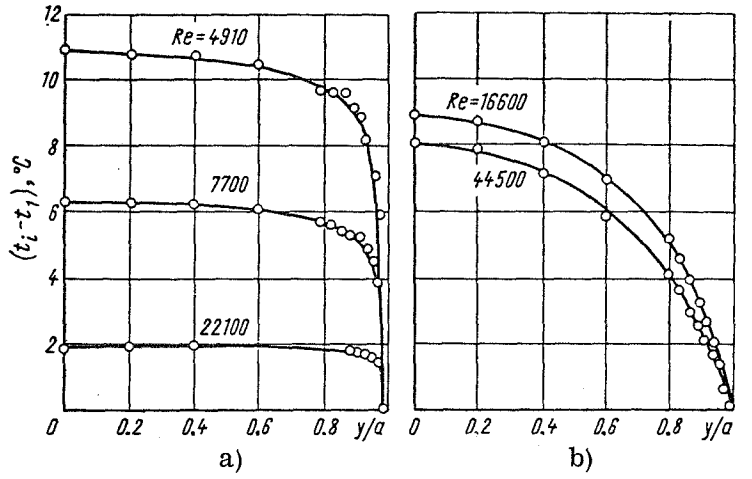


Figure 5. Temperature Distributions in a Square Duct for x/a = 0 and for Heating From Four Sides for Water (a) and for Mercury (b); t_1 is the Temperature of the Fluid When y/a = 0.99.

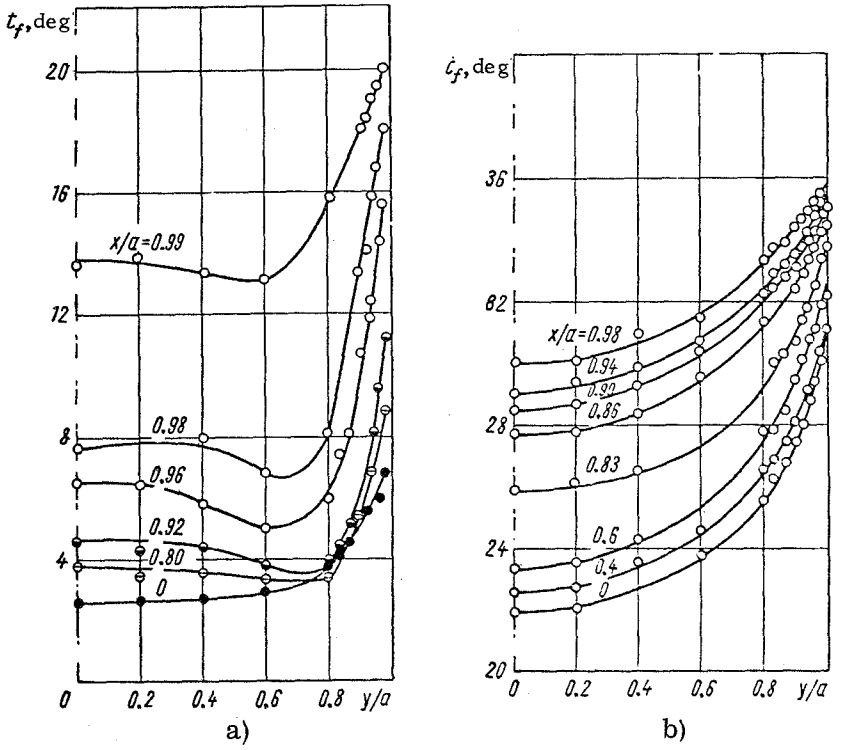


Figure 6. Temperature Distributions in Different Flow Cross Sections for Heating From Four Sides for Water (a) (Re = 4910) and Mercury (b) (Re = 16,600).

reduction in the slope of the temperature curve at x/a = 0 can be attributed to the effect of secondary flows, directed from the heat-conducting surface to the center of the duct [3]. As was shown experimentally, the secondary flows in a square duct occupy the entire duct cross section. The fluid is carried by the secondary flows along a diagonal from the center to the corner of the duct, and then the secondary flow flows along the wall and then from the center region of the wall to the center of the duct. When x/a = 0, the direction of the secondary flow coincides with the direction of the thermal flux. Consequently, the secondary flow in this point should assist in reducing the temperature gradient. The effect of the secondary flows on the temperature distribution in the flow of water is particularly noticeable by the distortion of isotherms in Figs. 7 and 8. All figures display the characteristic inflection of the isotherms in the direction of the duct's corner.

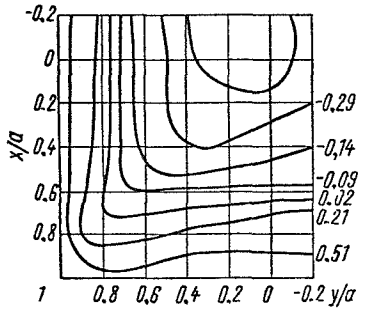


Figure 7. Isotherms in the Flow of Water in a Square Duct Heated From Four Sides for Re = 7700.

As can be seen from Fig. 9, the shape of isotherms in the flow of mercury differs appreciably from the shape of isotherms obtained in the flow of water. It is obvious that for such highly-conducting fluids as mercury, the effect of secondary flows on the shape of temperature fields is much smaller. When the heat is supplied asymmetrically (see Figs. 8 and 9) the temperature distribution in the fluid flow is affected by heat leakages along the heat conducting walls of the duct. The effective area of heat transfer is found to be larger and is not equal to the area of the heater. The behavior of isotherms near the unheated wall points to the fact that a part of /93
the heat supplied by the heater flows along the conducting walls and is transferred from them to the flow of the fluid. The quantity of heat which is transferred to the fluid by the unheated walls, and the shape of the temperature fields depend on the heat transfer coefficient, dimensions of the duct, thermal conductivity and wall thickness. In particular, it follows from Fig. 9 that in the case of heating from one and two sides the temperature of the fluid in the corners of the duct is lower than the temperature in the middle part of the wall and, consequently, the fluid and the wall in the corners of the duct are not overheated.

The high thermal conductivity of mercury, secondary flows and thermal fluxes through the wall result in the fact that the maximum temperature of the heat conducting surface is situated not in the corner of the duct, but in the center of the duct wall. Such a temperature distribution naturally depends on the boundary conditions at the outer surface. If the heater is wider than the width of the flow cross section of the duct and the heat transfer at the outer surface is nonuniform, for example, high heat transfer density in the corner of the duct, /94
then the temperature maximum of the heat conducting surface may be found to be in the corner of the duct.

We wish to note the following: in a square duct heated from four sides the isotherm corresponding to the average mixed temperature of mercury in a given

cross section is situated at a distance y/a and x/a $\sim$ 0.7, which coincides with the location of a similar isotherm in a pipe (for liquid metals).

Statistical Characteristics of Turbulent Temperature Fluctuations

Figure 10 shows oscillograms of temperature fluctuations in the flow of water in a duct heated from four sides with a specific thermal flux of q $\approx$ 15,000 watts/m^2 and Re = 4910. To the left of each oscillogram is shown the temperature scale. To the right are given the coordinates of the probe and its location is noted by a dot. It can be seen from the oscillogram that the characteristics of the fluctuation curve differ from one another in the different points of the duct's cross section. First, while in the central zone of the duct one observes high-frequency temperature fluctuations with a relative moderate amplitude ($\sim$1° C) (see Fig. 10a), /95 then in the region adjoining to the duct wall (x/a = 0, y/a = 0.98), the temperature fluctuations are as high as 10° C and their frequency is lower. In the very corner of the duct the fluctuation frequency is even lower (see Fig. 10c). Secondly, the oscillogram (see Fig. 10c) does not exhibit high-frequency components of the spectrum. This shows that only large-scale fluctuations are capable of entering the corner region. And, finally, the oscillograms in the corner of the duct are not symmetrical with respect to the time-average temperature in the given point of the flow, which can be seen from Fig. 10 and from other oscillograms which were obtained with higher water flow rates. The same asymmetry has been observed also in experiments with mercury. It consists in the fact that the temperature curve has peaks directed primarily in one direction away from the average temperature. For the region near the wall they are directed in the direction of smaller temperatures, while for the center region they are directed in the direction of higher temperatures. Apparently these peaks are produced when the center of a turbulent disturbances passes through the thermocouple. If the thermocouple is situated in a region adjoining the wall, then the peaks are produced by cold portions of the fluid which arrives from the center of the duct. In the central region of the flow the temperature peaks are brought about by the /96 passing of hot portions of the fluid, carried from the wall to the center of the flow through the thermocouple junction.

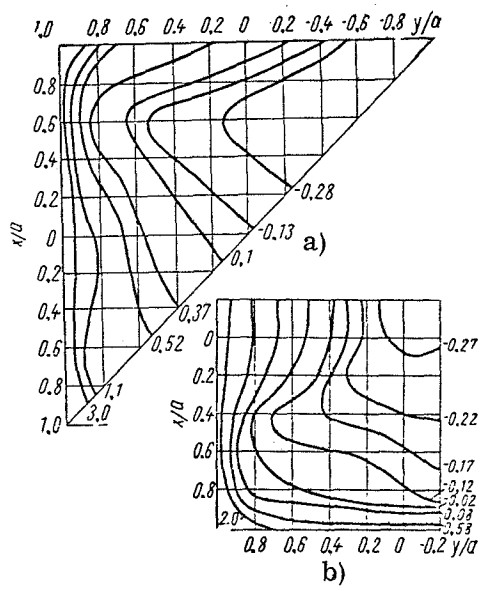


Figure 8. Isotherms in the Flow of Water in a Square Duct. a) Heating the Corner, Re = 4910; b) Heating From Two Sides (the Left Wall is Heated), Re = 4910.

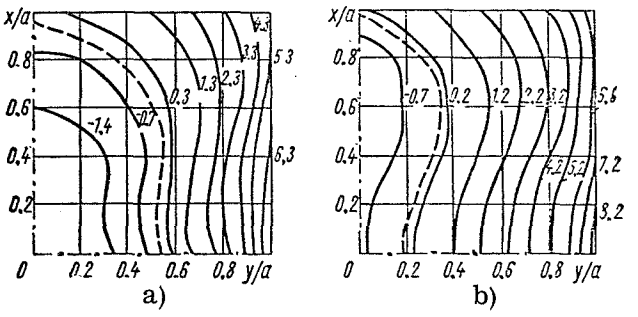


Figure 9. Isotherms in the Flow of Mercury in a Square Duct. a) Heating From Two Sides, Re = 28,400; b) Heating From One Side; Re = 28,200.

A more complete picture of the distribution of temperature fluctuations in different cross sections of the square duct was obtained using the correlometer. The results obtained by measuring the root-mean-square values of temperature fluctuations in water flowing in a square duct are shown in Figs. 11-13a. Figure 11 shows the curve of σ_t, the root-mean-square values of temperature fluctuations along lines parallel to axis Y. The parameter of these curves is the distance from the center of the duct along axis X.

The distribution of σ_t over the cross section of a square duct is shown more graphically in Fig. 12. The lines of this figure correspond to the locus of points with a constant root-mean-square temperature fluctuation of the flow. The parameter of curves in Fig. 12 is σ_t. As can be seen from Figs. 11 and 12, the distribution of σ_t over the duct cross section is quite complex. In the center part and at the walls σ_t has a minimum; the maxima are situated at the axial lines of the square at a distance of $\sim$14.5 mm (y/a = 0.97) from the center of the duct. At the wall σ_t increases very sharply. As x/a becomes larger (see Fig. 11) the maxima are reduced smoothly and move away from the wall of the duct. They move farthest away in the corner region. In the center region of the flow, when x/a and y/a $\leq$ 0.5, the magnitudes of σ_t are practically constant. In regions with 0.5 < x/a < 0.9 and 0.5 < y/a < 0.9 the curves of constant values of σ_t have a peculiar shape: at the axial lines these curves are displaced toward the center of the duct, while in the corner region they are extended along the bisectrix into the corner of the duct.

The shape of curves σ_t = const attests to the presence of secondary flows in the square duct. Let us examine the distribution of σ_t along the line y/a = = 0.8. The lines of σ_t = const for x/a = 0.6 are displaced to the center of the duct. The location of these displacements is the same as the maximum value of the normal velocity component of the secondary flow [3]. Lines σ_t = 0.7 and 0.9 for y/a = 0.3 conversely, are displaced toward the wall of the duct. The region of minimal σ_t for x/a $\approx$ 0.8 is identical with the location of the eddy of the secondary flow, where the normal components of the secondary flow velocity are at minimum. We thus notice a correspondence between the magnitudes of normal components of secondary flows and the magnitude of σ_t. /97

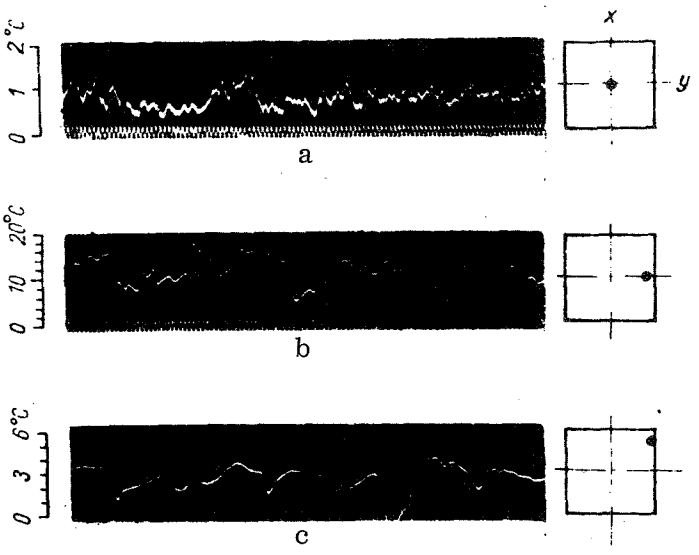


Figure 10. Oscillograms of Temperature Fluctuations in Water Flowing in a Square Duct. a) x = 0, y = 0; b) x = 0, y = 14.75 mm; c) x = 13 mm; y = 14.4 mm; Time Scale is 1 sec.

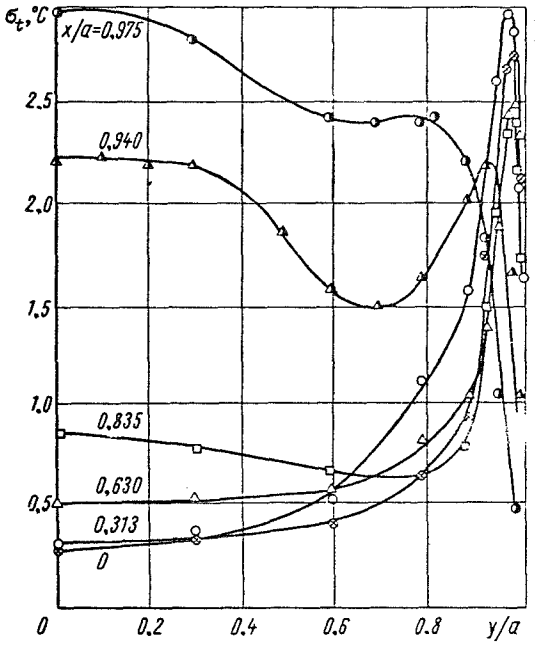


Figure 11. Curves of σ_t the Root-Mean-Square Values of Temperature Fluctuations in Different Cross Section of a Flow of Water, R=4910.

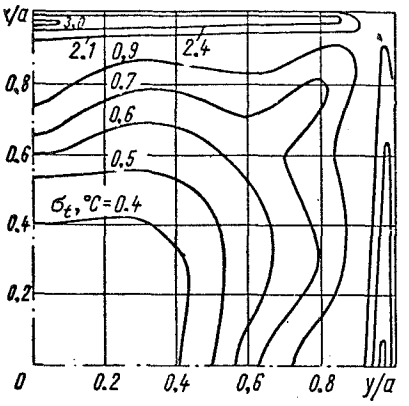
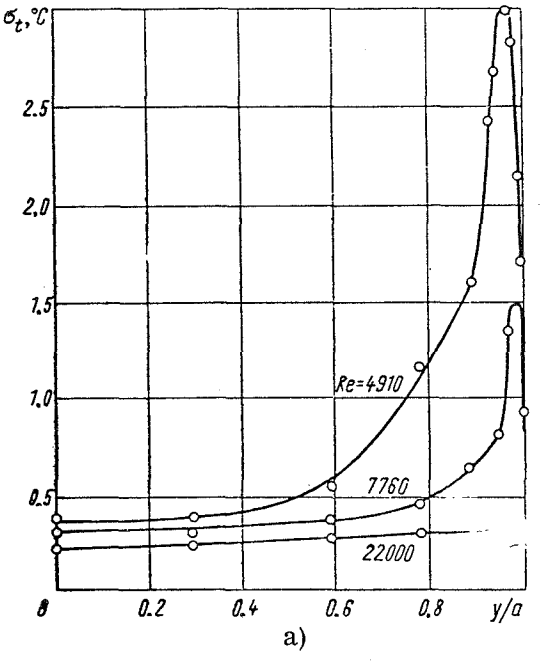
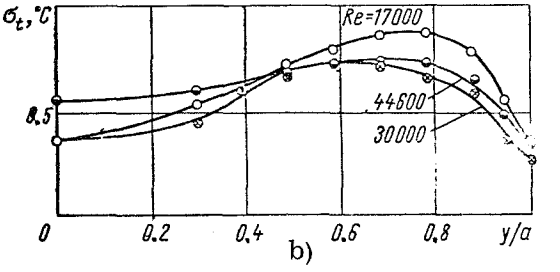


Figure 12. Distribution of Curves of Constant σ_t in the Cross Section of a Duct in the Flow of Water, Re = 4910.



a)



b)

Figure 13. Distributions of σ_t in a Square Duct Heated From Four Sides With a Flow of Water (a) and Mercury (b) for Different Reynolds Numbers.

Figure 13 shows the change in σ_t (determined at the same thermal flux) along the center line of the duct (x/a = 0) for different flow rates of water and mercury in a square duct. As Re is increased from 4910 to 22,000, the central region in which σ_t is practically constant, extends from y/a = 0.5 to y/a = 0.9 and the magnitude of σ_t becomes increasingly uniform over the duct's cross-section. The shape of the curves is no longer self-similar. The absence of a peak in the σ_t curve in the region near the wall at Re = 22,100 is produced by the relatively large dimensions of the pickup. The maximum of σ_t is located near the laminar sublayer, the thickness of which at the above Reynolds number is about 0.1 mm. Hence a thermocouple with an outer diameter of 0.25 mm is situated outside the region of maximum values of σ_t and, in addition, may appreciabley distort these readings.

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We wish to note that Figs. 11-13 display only the amplitude characteristics of the fluctuations. The frequency characteristics of the fluctuations, as was already shown, differ in different regions in the duct. Judging by the oscillograms, the closer the point to the wall and to the corner of the duct, the greater the contribution of low-frequency components. The average frequency of fluctuations, determined from the oscillograms of Fig. 10 as half of the number

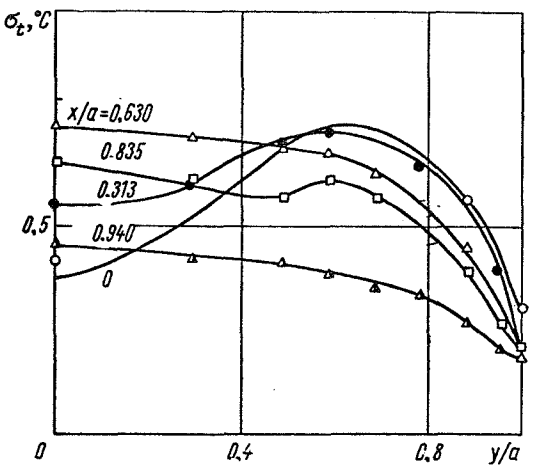

Figure 14. Distributions of the Root-Mean-Square Value of σ_t, the Temperature Fluctuation, in Different Cross Sections in a Flow of Mercury, Re = 30,400.

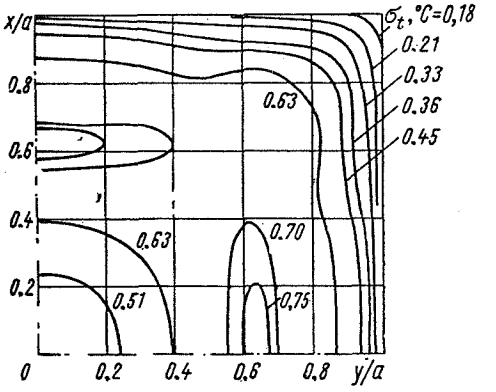

Figure 15. Distribution of Lines σ_t = const in the Cross Section of a Square Duct for Mercury, Re = 30,400.

of extrema of the temperature curve per second, was found to be 6.7, 2.8 and 0.9 cps for the central region, region near the wall and the duct's corner, respectively. The fluctuation characteristics for mercury were determined using the same equipment as for water.

Figure 14 shows values of σ_t along lines parallel to axis Y. Figure 15 shows lines of constant σ_t, constructed on the basis of curves of Fig. 14. It follows from these figures that the root-mean-square values of temperature fluctuations for mercury are distributed smoother than in experiments with water. The maximum of σ_t is also located on the axial lines of the cross section, but it is displaced appreciably from the wall of the duct to the center, i.e., /99
y for σ_{max} for Re = 30,400 is 9 mm (y/a = 0.6). This change in the distribution of σ_t with an increase in the thermal conductivity of the liquid [when water is replaced by mercury, DBL] agrees with a hypothesis stating that the amplitude of the turbulent temperature fluctuations is proportional to the product of the mixing path and the gradient of the average temperature [17]. In liquids with high thermal conductivities the temperature gradient exists in the entire interval from the wall to the center of the duct, while when heat is transferred to water the main temperature drop is observed near the wall. Hence, the magnitude of $\sigma_{t\,max}$, which is obtained in experiments with water is located at the wall of the duct, while that obtained with mercury is displaced to the center of the duct. The shape of curves of constant σ_t obtained in experiments with mercury differs from similar curves obtained with water (see Fig. 15). In the mercury experiments the effect of secondary flows on the σ_t field is also smaller, i.e., curves of σ_t = = 0.45, 0.36, etc., are extended in the direction of the duct's corner. This again

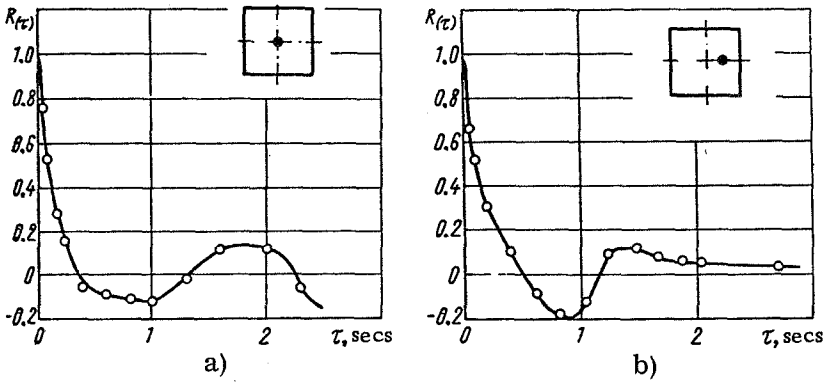

Figure 16. Normed Autocorrelation Function in the Center of a Turbulent Flow of Water in a Square Duct Heated From Four Sides; Re = 4910; a) x/a = 0, y/a = 0; b) x/a = 0; y/a = 0.8.

confirms the fact that in liquid metals with high thermal conductivity the effect of secondary flows on the heat transfer becomes lower. The effect of the average velocity within the given limits of variation in Re does not have a substantial effect on the magnitude of σ_t (see Fig. 13b). Comparing Figs. 11 and 13 we note that the magnitudes of σ_t in the center of the duct in experiments with water and mercury are close to one another by absolute magnitude. The magnitude of σ_t was measured with a specific thermal flux of q = 15,000 watts/m^2 provided by heaters on heating from four sides. When the Reynolds number is increased the magnitude of σ_t over a large part of the duct cross section in the flow of mercury becomes greater than for water

The magnitude of σ_t and the average frequency of fluctuations do not fully /100 characterize the temperature fluctuations. An important characteristic is the probability that the fluid in the given point of the flow will have a given temperature. We have used the oscillograms to determined the distribution of the probability of temperature fluctuation amplitude, which is in good agreement with a normal distribution of random variables. Since the points examined are situated in different regions of the duct's cross section (for example, at the wall and at the corner of the duct) it may be assumed that in the first approximation the probability distribution of amplitudes of the temperature fluctuations conforms to the normal distribution for a large part of the cross section of the square duct.

Figures 16a and b show the normed autocorrelation function (correlation coefficient) in two points of a flow of water, calculated from the expression

$$R_\tau = \frac{t'(0) \cdot t(\tau)}{\overline{t'^2}}.$$

In the main body of the flow (see Fig. 16) the correlation coefficient decreases rapidly to zero; on further increase in τ, the time shift, function $R(\tau)$ fluctuates about zero and is gradually damped out. The life of the temperature disturbance, determined by the first intersection between the curve of R_τ and the abscissa axis is 0.35 and 0.55 secs. The life of a turbulent disturbances can be used to determine its dimensions. The characteristic dimension of a temperature disturbance in the axial direction is $L_1/a = \tau \cdot \bar{u}/a = 6$ (when Re = 4910), i.e., it is appreciably greater than the cross sectional dimensions of the duct.

Results

Our experimental work shows that for accurate calculation of temperature field in rectangular ducts it is insufficient to have relationships used for calculating heat transfer in round pipes. This is true for liquid metal, as well as for ordinary, low-thermal-conductivity, heat transfer agents. /101

The fact that secondary flows have an appreciable effect on the heat transfer process in a square duct was proven experimentally. This was demonstrated by the characteristic distribution of the wall temperature, by the specific shape of the mean temperature distributions and of the temperature fluctuation intensity in the fluid flow.

The existing semi-empirical theory of heat transfer do not at all take into account the contribution of secondary flows, hence, these theories should be corrected accordingly.

The heat transfer in a square duct, in addition to secondary flows, is also affected by heat fluxes through the wall, which depend on the duct geometry, thickness and thermal conductivity of the wall, which is a characteristic feature of all nonround channels. Hence, for accurate calculations of nonround ducts from the point of view of fluid flow as well as from the point of view of heat transfer one must examine each specific geometry and solve the problem on taking into account all the factors which affect the temperature field.

NOTATION

t_{f_1} temperature of fluid at the duct inlet, ° C

t_{w_1} temperature at the outer surface of the wall, ° C

t_{w_2} temperature at the inner surface of the wall, ° C

G flow rate of heat-transfer agent

σ_t root-mean-square value of temperature fluctuations, °C

x, y current coordinates (the coordinate origin is in the center of the duct)

a half-width of the flow cross section

b wall thickness

d_h hydraulic diameter of the duct

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